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Energy Consumption of Heavy Road Vehicles: Field Parameter Measurements and Cross-Impact Analysis Summary Report

Prepared for the
Transportation Development Centre
Policy and Coordination Group
Transport Canada

April 1989

by

D.A. Ljubic
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Y. Provencher, F.E., M.Sc.

Forest Engineering Research Institute of Canada

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16. Abstract

This report summarizes the results of three years' research on factors which influence the energy consumption of heavy road vehicles. The relevant factors are grouped into a system called SICMER (system of interrelations among driver, machine, environment, and road). Certain relationships have become evident from the analysis of this system. These relationships touch on several aspects: energy transformation in the engine, energy transmission to the wheels, energy balance between tires and road, complex movement of articulated vehicles, and the influence of the driver's technique. Each of these aspects has been studied in its natural environment. Some comparisons of fuel consumption are presented in the following cases: differences in drivers' techniques; different combinations of the "semi-trailer undercarriage"; and different road surfaces and profiles. Prototypes of drivers' assistance instruments have been designed based on some of the relationships studied.

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16. Résumé

Ce rapport contient les résultats de trois années de recherche dans l'étude des facteurs qui influencent la consommation énergétique des véhicules routiers lourds. Les différents facteurs qui interviennent sont regroupés en un système d'interrelations, appelé SICMER (Système d'Interrelations entre le Chauffeur, la Machine, l'Environnement et la Route). L'analyse du système ainsi défini a permis de dégager certains mécanismes de relations. Ces relations portent sur plusieurs aspects, à savoir: la transformation énergétique dans le moteur, la transmission de l'énergie aux roues, l'équilibre énergétique entre les pneus et le sol, le mouvement complexe d'un véhicule articulé et l'influence de la technique de conduite. Chacun de ces aspects est étudié dans son environnement naturel. Certaines comparaisons entre consommation de carburant sont présentées dans les cas suivants: variation de la technique de conduite, différentes combinaisons du train roulant de la remorque, différentes surfaces et différents profils de route. Des prototypes d'instruments d'aide à la conduite sont conçus sur la base de certaines relations étudiées.

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1. INTRODUCTION

This report, titled "Energy consumption of heavy road vehicles: field parameter measurements and cross-impact analysis", is the logical continuation of efforts by the Forest Engineering Research Institute of Canada (FERIC) to examine, quantify and correlate the factors which influence the energy consumption, wear and productivity of trucking rigs.

In this era of phenomenal technological development, the overall efficiency of rolling stock ranges between 20 and 30% of the energy potential used as fuel. Clearly, much remains to be done in the national and international interest to upgrade the performance of our modes of transportation as regards both production and use. Needless to say, the improvement of vehicle efficiency is largely contingent upon improvements at the manufacturing level. But progress in that area is halting given that manufacturers are interested mainly in increasing profits, not reducing sales.

Furthermore, any improvement in vehicle efficiency (reduction in energy consumption) triggers a reduction in environmental pollution.

For many years, the operation of rolling stock was geared exclusively to productivity, regardless of the costs incurred through excessive fuel consumption, machine wear or driving practices aimed solely at production. All of these factors combined have played a part in putting transportation costs at 30 to 50% of total production costs. In Canada, a vast country that relies heavily on road transport, any reduction in energy consumption translates into a cut in the cost of transportation and thus in cost price. It is well-known that the quality of performance of a trucking rig depends not only on its basic characteristics (e.g., design, components, assembly, specifications and materials) but also on the way it is used under various operating conditions.

Improving the productivity of transport vehicles requires optimization of their design and specifications, enabling suppliers to deliver rigs that are better suited to our operating conditions. It also calls for measures to optimize their utilization. Methods must therefore be developed for the theoretical and experimental field study of trucking rigs, thereby laying the ground for analysis of the parameters comprising the System of Interaction between Driver, Machine, Environment and Road (SICMER). Analysis of this system should provide for improved transport cycle design and planning, lower costs and higher productivity.

The principal results of this analysis are presented here. For a more detailed treatment, consult the corresponding sections of report TP 9715F (published in French only).

2. AIMS AND METHODS

This project was aimed chiefly at the field analysis of driver-machine-environment-road interactions by means of a test truck and trailer. This entailed automatic, simultaneous logging of the key variables on the road, allowing for the exploration of overall differences in energy consumption or for instantaneous measurement of the variables.

The secondary aims were to explore driving techniques by monitoring 35 drivers manoeuvring the same trucking rig (tractor, trailer and load) on the same road; to correlate energy consumption and road curves, grades and surfaces; and to investigate the impact of selected components of the test trailer undercarriage on energy consumption.

With a view to analysing driver-machine-environment-road interactions, FERIC developed a theoretical method that uses data logged in the field as the input. Once validated, this method served as the basis for analysing the cross-impacts between various factors which influence energy consumption.

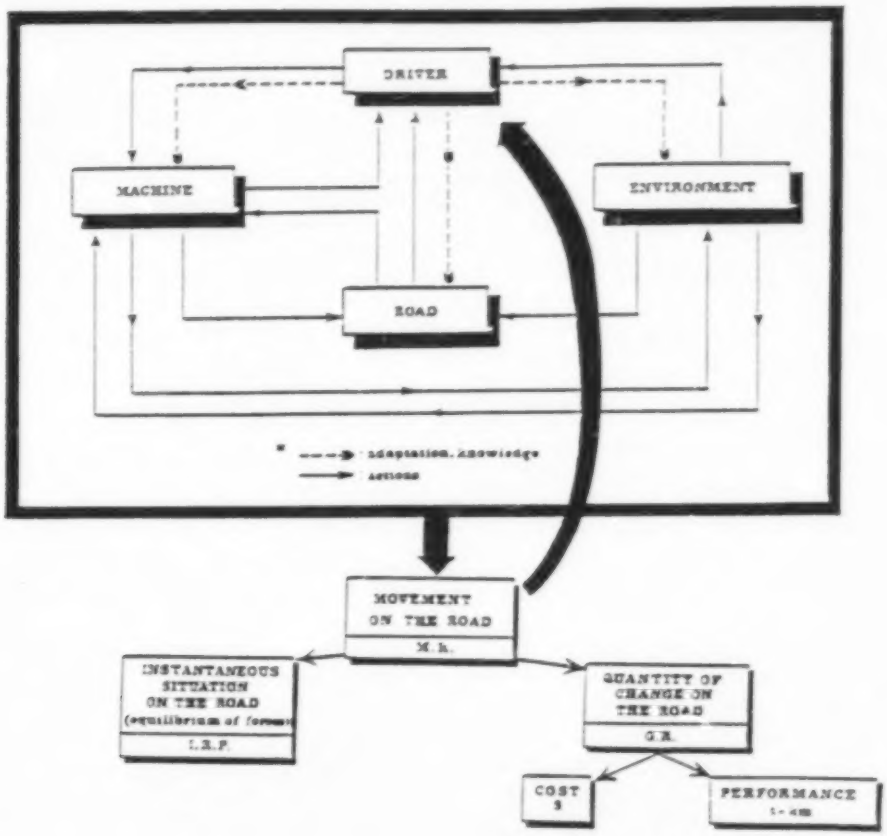

Figure 1 Model for analysis of the System of Interaction between Driver, Machine, Environment and Road (SICMER)

3. THEORETICAL AND EXPERIMENTAL DEVELOPMENT OF STUDY METHODS

A transport cycle productivity and cost analysis based on the latest automatic field datalogging techniques taking account of the performance potential of the articulated rig, as well as the road, the environment and the driver's actions, requires the development of complex theoretical and experimental methods for dynamic, instantaneous correlation of the variables and parameters involved. The aim is rapid field assessment of the complex cross-impacts between the driver's actions, energy conversion in the engine, energy transmission along the drive train, the laws governing tire-ground contact, and equilibrium of the forces and moments acting on the vehicle under typical operating conditions (e.g., on a curving upgrade).

Prior to such analyses, methods to determine energy conversion within the engine had to be developed. To make this determination, energy potential in the form of consumed fuel can be measured at the engine intake and then correlated by means of field data with measurements at the engine output, with particular attention to engine torque and rotating velocity measurements. In addition, the consumption and torque data were correlated with the driver's pedal manoeuvres for a given transmission ratio.

Figure 2 diagrams the means of measuring torque at the transmission output, as well as pedal depression, fuel consumption and engine rotating velocity.

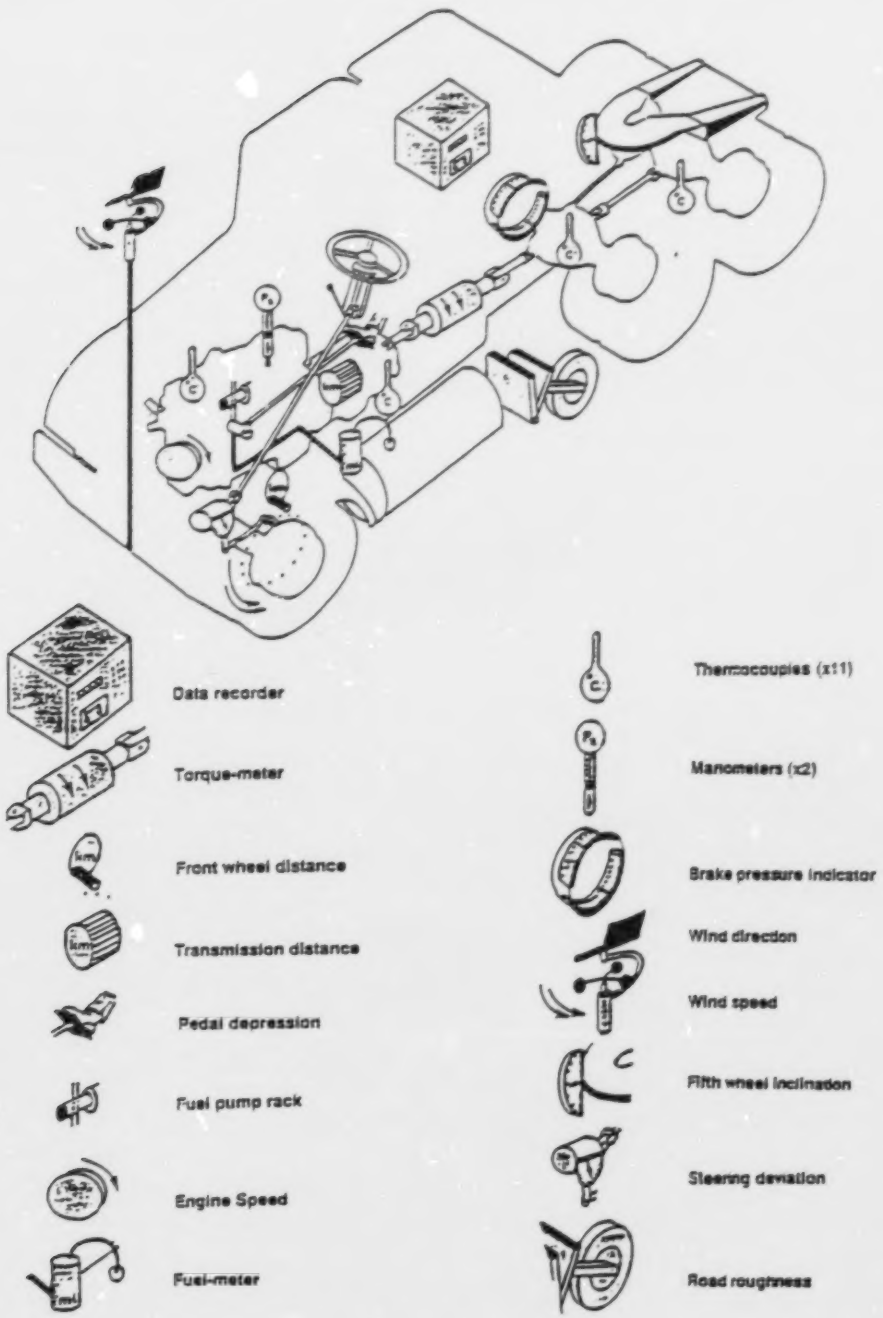

Figure 2 Diagram of datalogging system

The torque measured by torquemeter is reduced to the engine through division by the transmission ratio. It satisfies the following equation:

$$M_{CMM} = \frac{M_{CM}}{i_{TR}} = M_E + \frac{M_{TR}}{i_{TR}} = M_{IN} + M_{RPM} + M_{IRM} + M_{ACC} + \frac{M_{TR}}{i_{TR}} \qquad (4.1)$$

where:

M_{CMM} - dynamic moment measured by torquemeter and reduced to engine (N·m)

M_{CM} - torque measured by torquemeter (N·m)

i_{TR} - transmission ratio

M_E - effective moment at engine output (N·m)

M_{TR} - moment representing loss through transmission[1] (N·m)

M_{IN} - indicated moment of engine (N·m)

M_{RPM} - moment of passive engine resistances (N·m)

M_{IRM} - moment due to rotating inertias of engine (N·m)

M_{ACC} - moment due to engine accessories (N·m)

In this equation, M_{RPM}, M_{TR} and M_{ACC} are always negative. M_{IRM} may be negative ($\frac{d\omega}{dt} > 0$), nil ($\frac{d\omega}{dt} = 0$) or positive ($\frac{d\omega}{dt} < 0$). The indicated engine moment M_{IN} is always positive.

Figure 3 relates selected pedal depressions to the torque measured by torquemeter and reduced to the engine by equation 4.1 as a function of engine rotating velocity. The curves are plotted entirely from data logged in the field under widely varying conditions.

These curves were validated by comparing the torques measured at full load with the torque curves provided by the engine manufacturer. Figure 4 illustrates this comparison, showing a close match between the measurements made in the field and on a manufacturing test-bench (disregarding losses through the transmission).

(1) Taking account of the transmission moment of inertia.

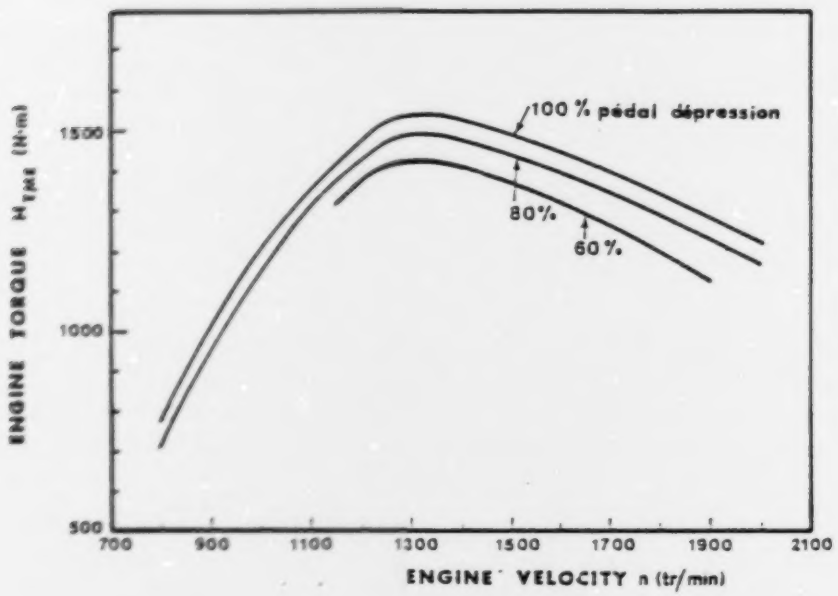

Figure 3 Moment of torque measured by torquemeter
and reduced to the engine as a function of engine rotating velocity

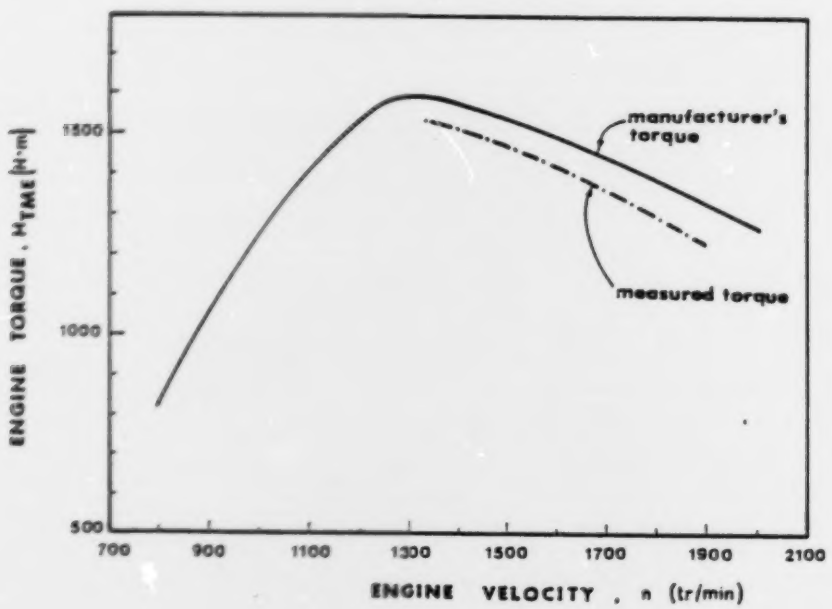

Figure 4 Comparison of moments of engine torque (a) provided by the
manufacturer and (b) measured by torquemeter and reduced to the engine

Figure 5, which was also derived from field data, correlates specific consumption, expressed as kilograms (kg) of fuel per kilowatthour (kW/h) as determined by engine rotating velocity, with the test engine developing 261 kW at 2100 RPM and with selected pedal depressions. The field data served not only to determine energy conversion in the engine but also to correlate engine performance with pedal depression for a given transmission ratio.

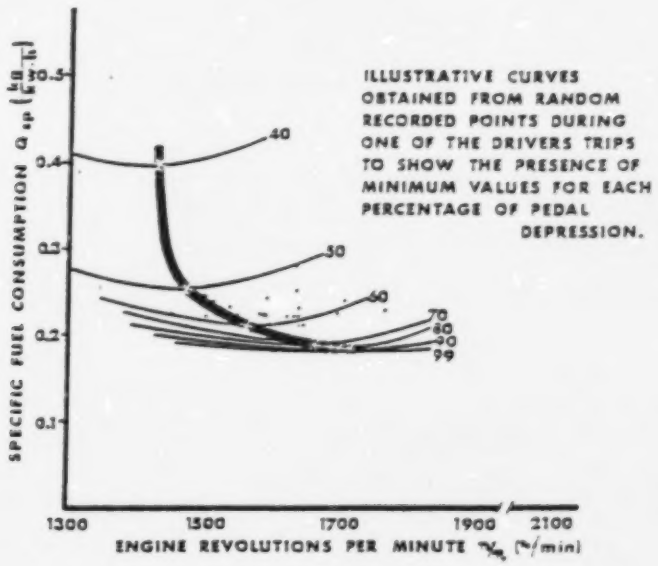

Figure 5 Specific consumption as determined by the rotating velocity of a Cummins Formula 350 NTC (III) engine for selected iso-pedals

A "mini-maxi" fuel pump regulator mounted on the first test engine was used to establish a direct link between engine performance and the driver's actions. The second engine on the test truck had an "all-state" regulator for the continuous adjustment of pedal depression. The first step was therefore to identify the link between the pedal and the fuel pump rack as a function of the different engine rotating velocities (see Fig. 6). Next, a correlation was drawn between the rack and the engine torque at different engine rotating velocities (see Fig. 7). In an effort to determine how any given parameter down from the engine affects fuel consumption, correlations were drawn between fuel consumption and torque at the engine output for various rotating velocities. This determination was made by torquemeter or by reducing the forces of resistance of the truck's traction wheels to the engine. Once validated, these cross-impacts are basic to the energy analysis of moving vehicles. This was the approach used to correlate energy conversion and the driver's actions for each engine. Knowledge of these baseline conditions is vital to all field energy analyses.

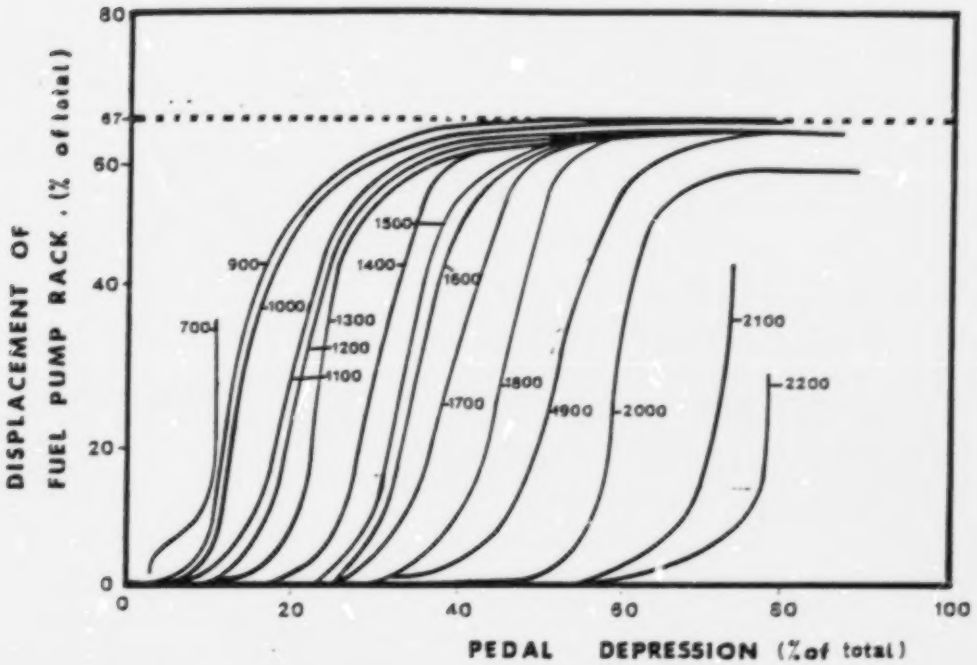

Figure 6 Correlation between pedal depression π and fuel pump rack displacement q

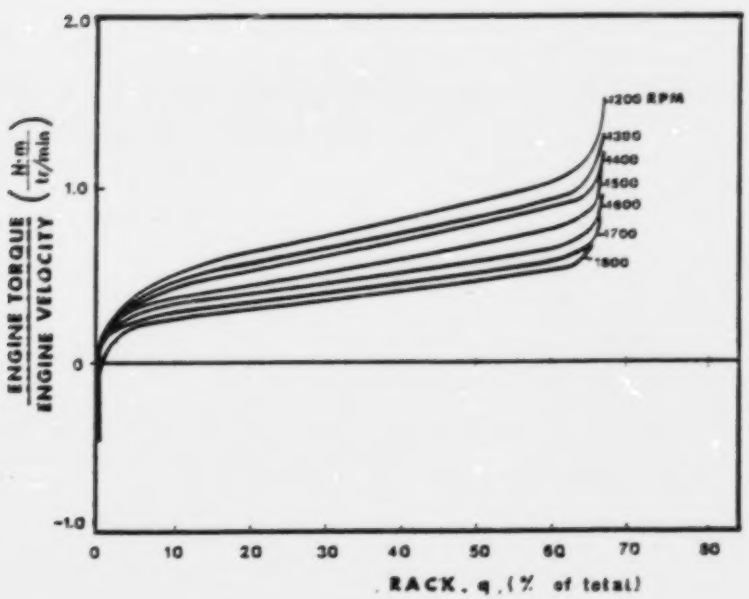

Figure 7 Correlation between rack, engine rotating velocity and torque at the engine output

To know the tractive force at the truck's traction wheels at any given instant, specific methods were required to determine the losses through the power transmission system. This previously prompted the development of theoretical and experimental methods to determine losses due to oil churning in the transmission and the two rear axles by means of free-wheeling deceleration tests. Constant-speed tests on a dynamometer or on the road are used to determine losses due to friction. Knowing the losses through the drive train, it becomes possible to determine at any instant the tractive force at the traction wheels. Dividing the tractive force at the traction wheels by the vehicle weight yields the dynamic traction factor. Figure 8 gives the dynamic factor as a function of road speed for the different transmission ratios and three pedal depressions. These determinations fulfill the second prerequisite for energy analyses.

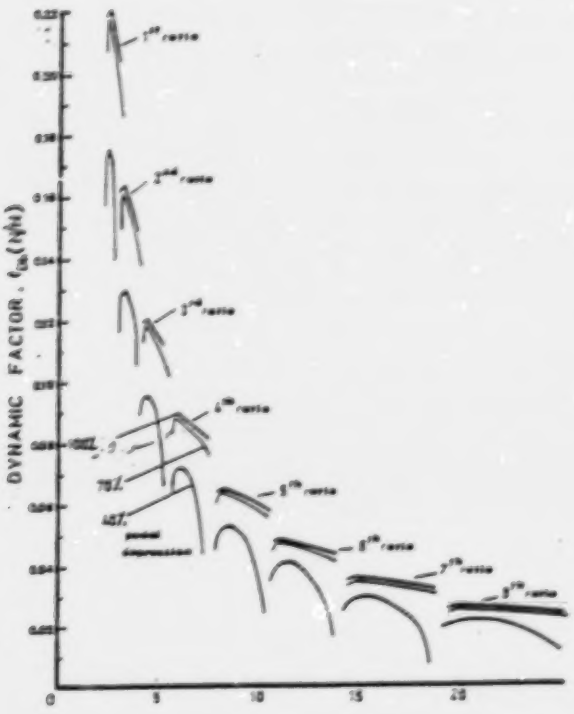

Figure 8 Diagram of dynamic factor as a function of road speed through all transmission ratios

Using the correlations derived from the field data to study the transition from pedal to engine and from engine to traction wheels, we matched the data logged at four points along the test road between Chénéville and Montpellier (see Fig. 9) against the readings from the sequential diagrams

representing the transition from pedal to traction wheels. The differences in fuel consumption at those four points--consumption measured by flowmeter as opposed to consumption calculated from the diagrams--come to 1.3% at point A, 0.3% at point B, 4.3% at point C and 5.8% at point D. Taking all the intervening factors into account, these figures show that the calculated correlations are accurate enough for use of the method.

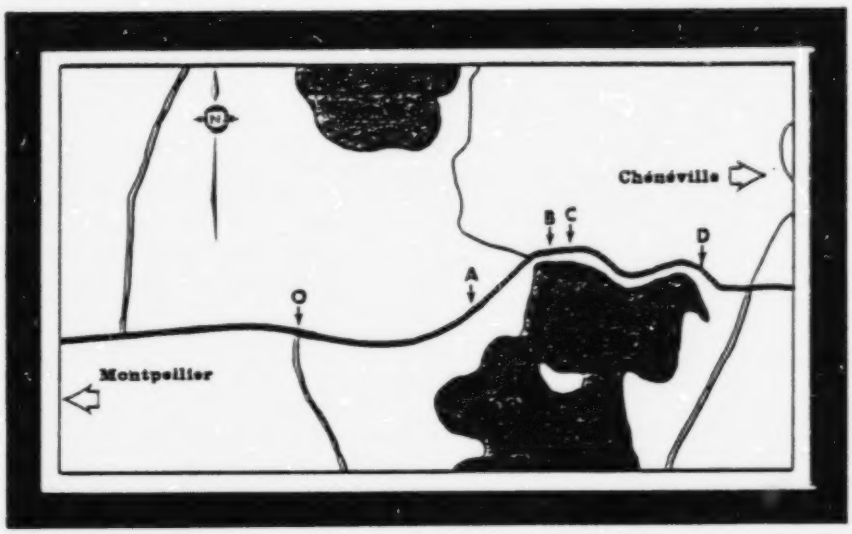

Figure 9 Test section of the Montpellier-Chénéville road

The nomogram in Figure 10 suggests the possibilities for analysis of driver-machine-road interactions in both directions. Once read into a computer routine, the curve equations plotted on the nomogram will be used to project the performances and energy consumption of a trucking rig equipped with the same engine as the test truck but for which all the other parameters are different.

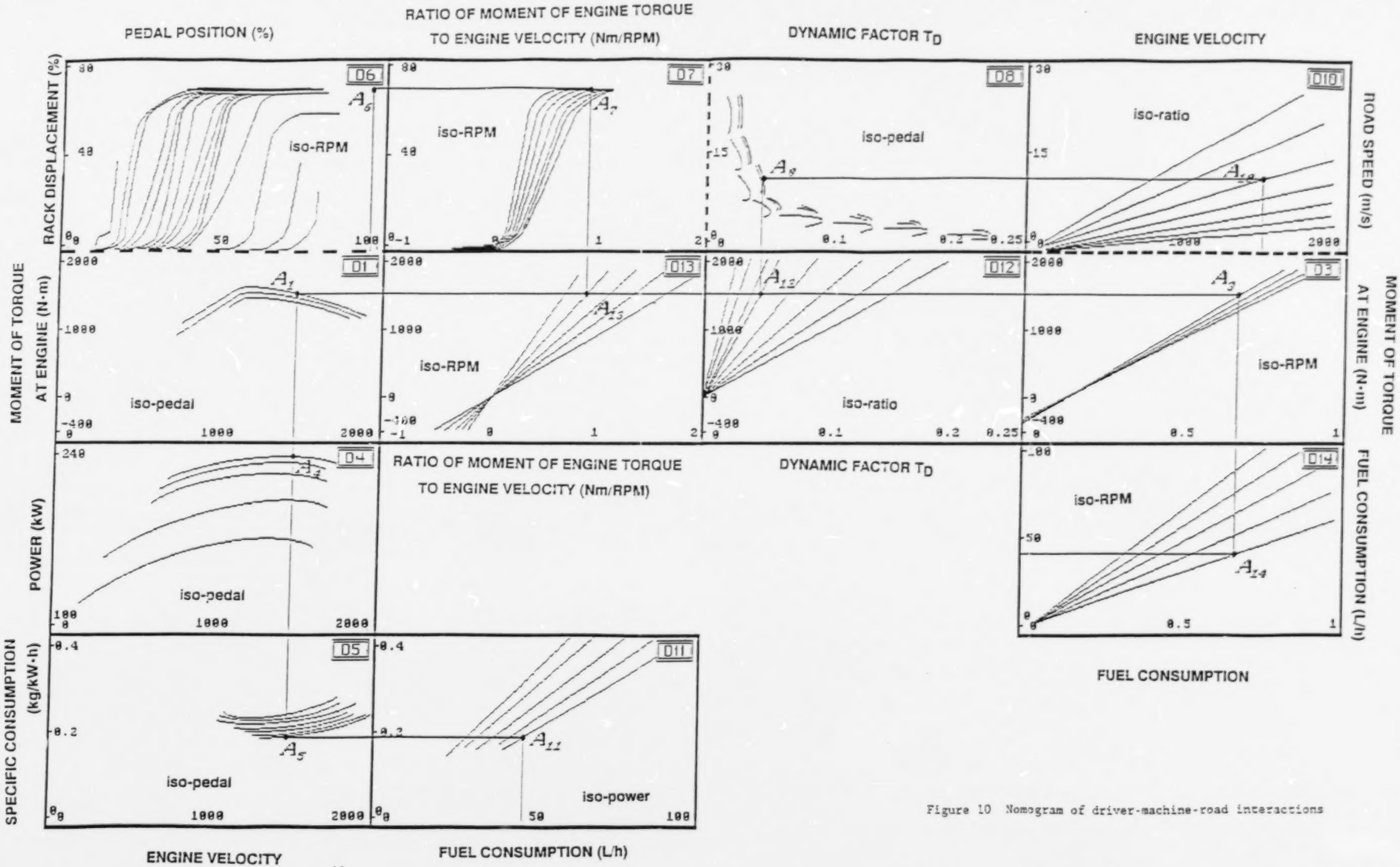

Figure 10 Nomogram of driver-machine-road interactions

Knowing the force applied between the tires and the ground at any given instant during dynamic movement on the road allows for examination of the instantaneous equilibrium of the exerted forces or moments.

The equation below expresses this equilibrium in its most general form.

$$F_{TDR} + F_{RA} + F_{RT} + F_I + F_{IR} + F_{IM} + F_{FRP} + F_{INC} = 0$$

with

$$F_{RT} = F_{RR}^{\circ} + F_M + F_{CRB} + F_{\mu G} + F_{RES}$$

where:

F_{TDR} - total dynamic force of traction at the wheels (N)

F_{RA} - dynamic force of air resistance (N)

F_{RR}° - force of rolling resistance on an ideally level, straight road without application of torque. The exponent "°" indicates near-zero torque (N).

These two forces are determined by free-wheeling deceleration on a straight, practically horizontal section of road.

F_M - complementary force of resistance due to application of torque (N)

F_{CRB} - force of resistance to forward motion due to curves in road (N)

$F_{\mu G}$ - force of resistance represented by energy loss due to slippage between traction wheels and ground (N)

F_{RES} - force of resistance to forward motion representing all residual resistances due to such factors as suspension, truck front-end configuration, locus of centre of gravity, truck and trailer axle spacing, etc. (N)

F_{RT} - total force of resistance to forward motion between tires and ground (N)

F_I - force of vehicle inertia (acceleration and deceleration) (N)

F_{IR} - force of inertia of rotating masses, i.e., of all rig wheels and drive train (N)

F_{IM} - force of engine inertia (N)

F_{FRP} - braking force (N)

F_{INC} - directional force of resistance due to vertical grade in section of road (N). Equals $g \cdot m_T \cdot \sin\alpha p$ where m_T is the mass of the rig in kg, αp is the slope in the road, considered positive for an upgrade and negative for a downgrade, and g is the ground acceleration in m/s^2.

F_{TDR} is the force exerted between the traction wheels of the truck and the ground. It is determined by the moment measured by torquemeter, reduced to the wheels, converted into force and reduced by the losses due to churning and friction in the two rear axles. $F_I + F_{IR} + F_{IM}$ are the forces of inertia calculated from the mass of the rig and the accelerations (axial in the case of straight movement and angular in the case of curving movement). F_{FRP} is calculated from the logged air pressure of the braking system. F_{INC} is calculated from the grade angles. F_M and $M_{\mu G}$ are calculated by equations specially developed in the report. To determine F_{CRB}, the force occurring in a curve, an equilibrium equation was developed to take account of the moments of resistance to rotation and tire side slip (see Appendix B of report). That particular equation was made into a computer routine. To verify the accuracy and repeatability of this method of calculation, the vertical grades of a 3000-m test section were calculatd by means of the same equation and then matched against the same grades derived with a theodolite. Figure 11 compares the two vertical profiles and shows satisfactory concurrence between them, given a mean absolute deviation of 0.5%. It also shows close proximity of the profiles, except at the three braking points.

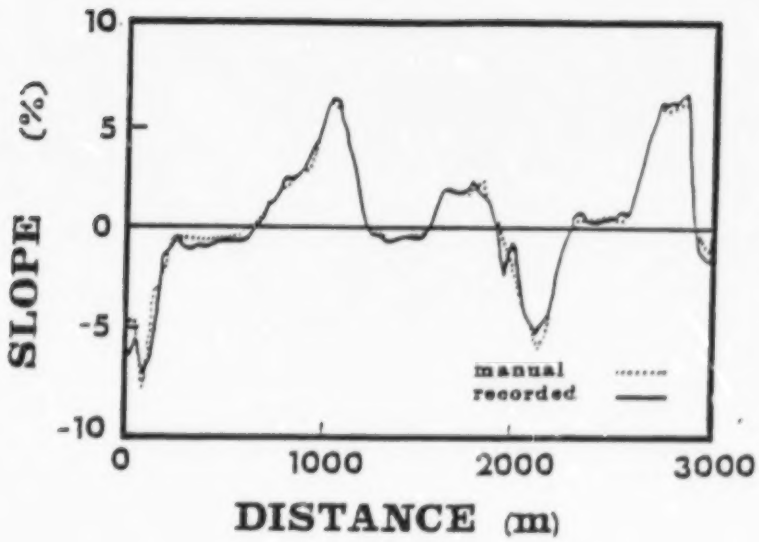

Figure 11 Comparison of grades
a) Curve obtained by theodolite
b) Curve derived from logged data

This provided general validation of the method which FERIC developed to calculate vehicle performance and energy consumption. This method was then used to calculate resistances and energy consumption upon the negotiation of curves and grades.

Comparison of the two profiles in Figure 11 points up the possibility of dynamic characterization of a given road by means of one or more truck runs on that road.

The test truck fitted with a torquemeter was used to correlate the supercharging pressure and the measured torque. Figure 12 compares the torque calculated from that pressure with the torquemeter measurement. The accuracy of the comparison appears to be acceptable. This principle was applied to design an instrument to indicate power and fuel consumption. That new indicator is not yet operational.

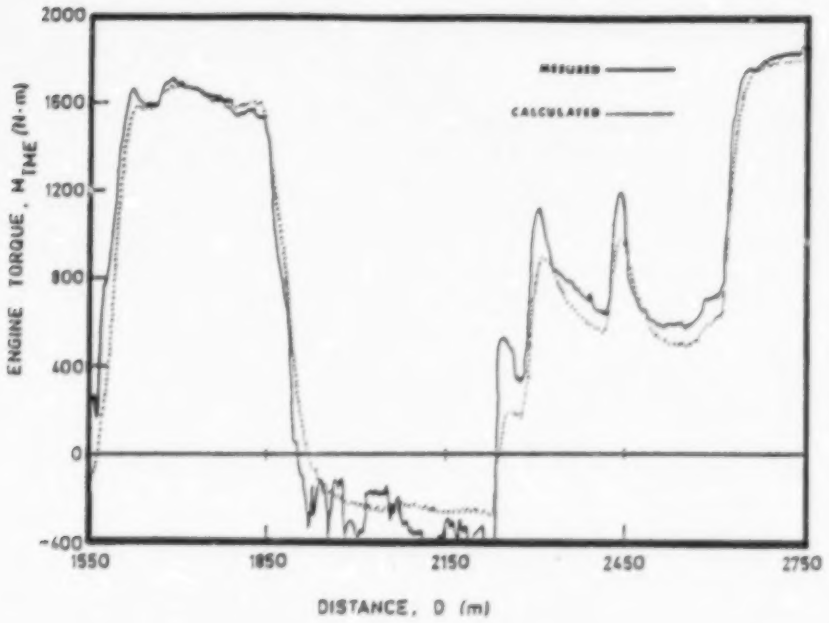

Figure 12 Curves of torque moment
a) measured by torquemeter
b) calculated from supercharging pressure

Having established and validated theoretical and experimental methods of calculating the performance and energy consumption of the driver-machine-environment-road system, it is possible to explore the experimental or calculated results, using the logged data as input.

4. EXPERIMENTAL RESULTS

Two types of tests were conducted for the analyses involved, namely:

- Analysis of driver-machine-environment-road interactions and their impact on energy consumption;
- analysis of the impact of trailer undercarriage type and configuration on energy consumption.

4.1 Analysis of driver-machine-environment-road interactions and impact on energy consumption

The tests required for analysis of the interactions between driver, machine, environment and road were conducted on a specially selected test run comprised of a section of highway 309 south of Mont-Laurier, a section of highway 311 up to Lac des Écorces, and from there to Mont-Laurier. Using that test run, the driving techniques of 35 professional drivers were logged with the same test truck (equipped with sensors to record the key variables), trailer and load.

The test run used by all the drivers was 86 km long, divided into two sections, the first 32 km long and the second 54 km long.

The two approaches described below were used to analyse the logged data.

4.1.1 Analysis of the dynamics of the driver-machine-environment-road system and driving techniques from instantaneous data logged at various points along the road

Let us look at an example of driver-machine-environment-road interactions on the "long island" section of the test road. Figure 13 gives horizontal and vertical profiles for this section. This example focuses more particularly on driver no. 20 negotiating the section between points A and F shown on the figure. It illustrates the driver's perception of the environment and his response, i.e., his actions on the machine, taking account of his driving technique. The test calls for negotiating a curving downgrade. Starting at point A, the driver descends a 3.8% hill, applies the engine brake and decelerates.

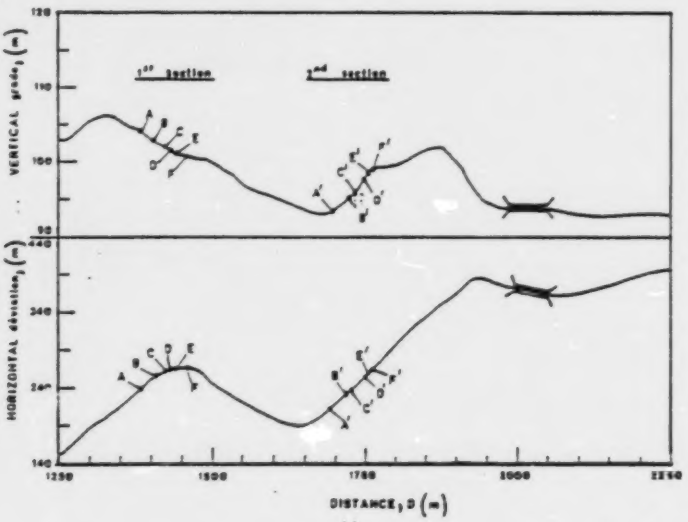

Figure 13 Horizontal and vertical profiles of the "long island" section of the road

Consumption is very low at the starting point (A) given that the rig is descending the hill with the engine engaged. This reduces internal engine friction somewhat and thus keeps consumption very low (2.52 L/h). At point B, the driver notices a curve and presses the brake pedal for a few seconds. This reduces the positive impact of the truck's thrust on the engine, and consumption starts rising, reaching 7.26 L/h at point C. At point C, in the curve, the driver shifts gear from 2.14:1 to 2.74:1. Between points C and D, consumption increases to 10.73 L/h, representing the fuel consumption of the engine idling at about 600 RPM. During this shift in gear, the road speed drops slightly, the pedal depression increases and the steering wheel deflection is reduced.

This road-driver scenario examines the transition from the driver's perception to his handling of the vehicle and thus to vehicle speed, torque (power) and consumption. Knowledge of driver-machine-environment-road interactions is vital to every energy analysis of a trucking rig.

Let us now compare the gearshifting techniques of three other drivers (nos. 17, 26 and 35) on the straight upgrade between 1650 and 1900 m (Fig. 14, points A" to E"). Looking at the gearshifting diagram for driver no. 17, he was in 8th gear (1.65:1) at point A". Realizing that the engine was starting to choke, he decided to downshift from 1.65:1 to 2.14:1. He changed gears rapidly and smoothly, i.e., without loss of road speed despite the steep upgrade. In addition, the driver kept the engine from choking by immediately downshifting and raising the rotating velocity above that of the maximum engine torque.

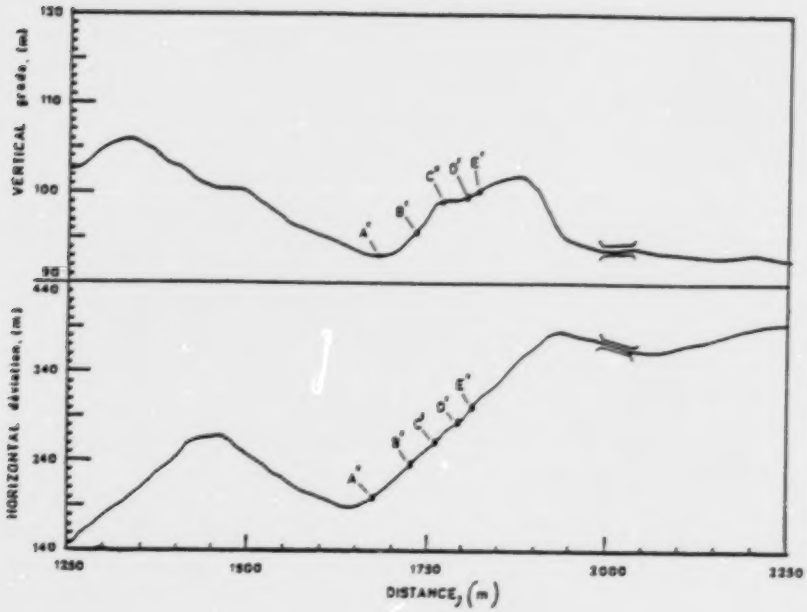

Figure 14 Horizontal and vertical profiles of the "long island" section

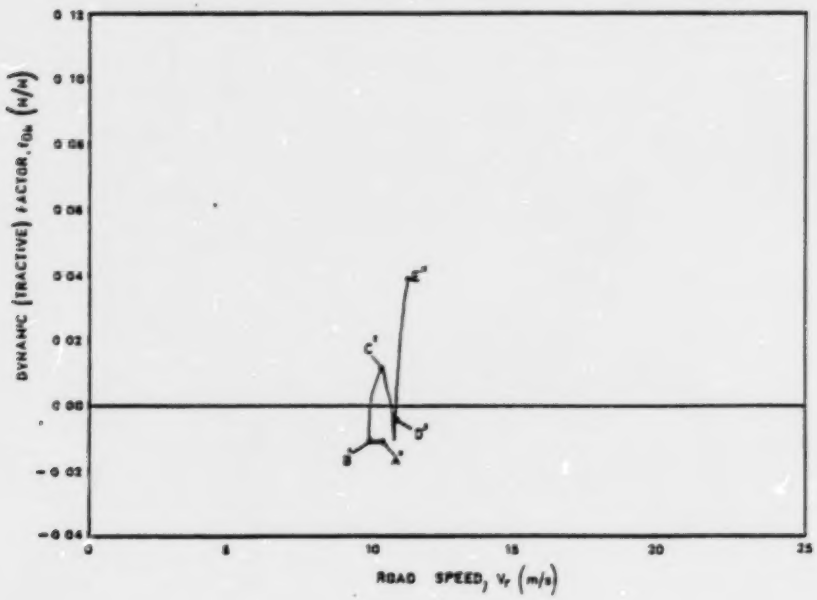

Figure 15 Diagram of the dynamic factor as determined by road speed, driver no. 17

Driver no. 26 let the road speed and engine velocity drop too low and was forced to stop on the hill. Actually, the engine dropped to 600 RPM, which meant it was idling. This is a typical example of poor driving technique on an upgrade.

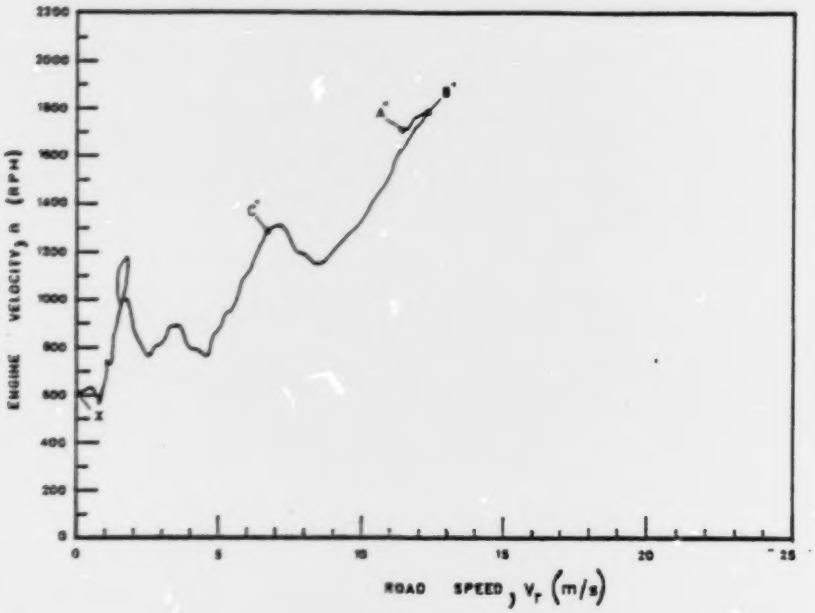

Figure 16 Gearshifting diagram for driver no. 26

Despite a slight reduction in road speed, driver no. 35 shifted gears smoothly, keeping the velocity of the engine above that of the maximum torque speed. His technique is very good.

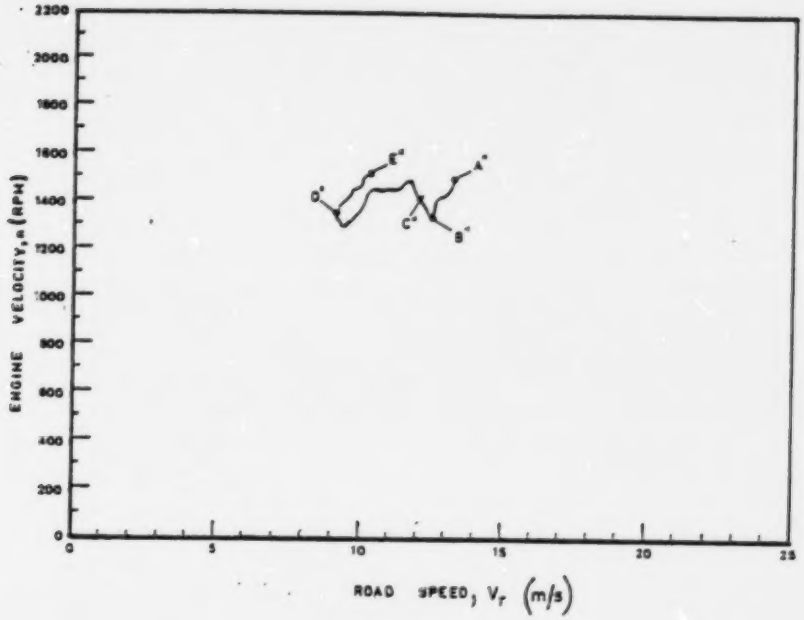

Figure 17 Gearshifting diagram for driver no. 35

These examples clearly illustrate the differences in gearshifting techniques when negotiating an upgrade.

The table below presents selected data logged for three drivers (nos. 17, 20 and 35) along this entire test section.

Table 1 Comparative means of selected variables for three drivers

Variable	Mean values		
	driver no.17	driver no.20	driver no.35
Time (s)	214	291	192
Engine velocity (RPM)	1564	1332	1435
Speed (m/s)	9.32	6.84	10.39
Torque (N•m)	1398	922	1034
Consumption (mL/m)	0.708	0.717	0.661
Power (kW)	113	61	101

This table shows a 7.8% difference in fuel consumption for driver no. 20 and driver no. 35, a difference stemming entirely from their different driving techniques.

This brief examination of the interactions between the SICMER components points up future possibilities for energy analyses. It was also an opportunity to study the driving techniques of the various drivers, providing vital information for future optimization of the transport cycle.

4.1.2 Analysis and comparison of driving techniques by means of cumulative mean data

Mean data are used here for a summary comparison of the driving techniques of the 35 drivers. Table 2 gives a idea of certain variables logged or calculated for each driver on the first, 32-km test section. As can be seen, the mean speed ranges between 60 and 79 km/h, depending on the driver. Power consumption ranges between 135 and 219 KW, or more than 1.5 times more. Fuel consumption is between 45.6 and 61 L/100 km, for a difference of 25%. Thus, all other things being equal, the difference in consumption for drivers nos. 26 and 35 is due solely to their respective driving techniques.

These differences show that drivers wield a tremendous influence on the productivity and costs of a transport company and should receive ongoing training coupled with incentives.

This report also provides a driver classification, analyses the correlations between variables, establishes a coefficient of driving performance and offers drivers some helpful advice on economical driving. The first part examines certain results obtained during the test runs south of Mont-Laurier; the second provides a detailed explanation of how drivers should go about improving their driving technique.

Table 2 Totals and means of logged or calculated variables for the first, 32-km road section, for the 35 drivers

driver no.	total distance (m)	total time (s)	mean engine velocity (RPM)	rate of consumption (mL/m)	rate of consumption (mL/RPM)	pedal depression (%)	mean torque	mean power positive	mean power negative
1	32245.3	1593.3	1632	0.532	0.396	42.820	980.7	183.34	-31.49
2	32281	1580.5	1581	0.538	0.417	48.452	959.2	191.27	-33.25
3	32320.9	1940.8	1597	0.497	0.311	33.020	911.8	134.23	-31.11
4	32152.3	1586.5	1653	0.582	0.428	46.429	1092.3	218.61	-35.49
5	32257	1637.7	1693	0.561	0.391	41.927	977	178.83	-36.94
6	32286.8	1776.8	1566	0.558	0.389	44.542	1018.8	177.25	-33.45
7	32249.4	1587.4	1699	0.582	0.418	43.647	*949.1	*177.82	*-36.02
8	32016	1738.8	1810	0.589	0.360	40.635	*949.1	*177.82	*-36.02
9	32388.4	1557.2	1670	0.556	0.416	59.358	*949.1	*177.82	*-36.02
10	32345.4	1692.3	1824	0.524	0.370	41.898	*949.1	*177.82	*-36.02
11	32367.9	1936.9	1438	0.500	0.349	35.109	*949.1	*177.82	*-36.02
12	32326.8	1760.4	1465	0.518	0.390	40.198	*949.1	*177.82	*-36.02
13	32274.6	1526.3	1578	0.486	0.391	39.585	*949.1	*177.82	*-36.02
14	32327.4	1532.5	1712	0.533	0.394	41.658	939.4	185.33	-39.74
15	32325.4	1653.9	1548	0.533	0.404	41.192	986	177.88	-31.89
16	32348.1	1808.2	1675	0.551	0.353	47.498	1021	181.11	-39
17	32271.4	1508.2	1696	0.485	0.367	54.702	1087.9	199.18	-29.09
18	32343.2	1580.8	1630	0.550	0.414	42.811	*949.1	*177.82	*-36.02
19	32423.7	1607	1709	0.551	0.390	39.011	973	179.42	-40.23
20	32248.8	1867.6	1546	0.505	0.339	31.340	838.2	135.28	-41.7
21	32305.6	1566.8	1592	0.519	0.403	42.307	943	171.89	-33.95
22	32250.1	1589.5	1578	0.513	0.398	39.825	901.6	165.65	-30.58
23	32244.6	1604.3	1661	0.462	0.336	40.563	884.2	167.69	-34.52
24	32304	1514.5	1794	0.518	0.370	42.923	896.4	172.69	-40.5
25	32309	1545.2	1660	0.507	0.383	46.618	911.5	179.18	-38.57
26	32286.9	1550	1676	0.610	0.454	61.778	1108.4	208.98	-34.69
27	32402.5	1556	1657	0.557	0.420	45.457	988.1	193.46	-36.16
28	32315	1613	1721	0.557	0.389	47.047	936.3	164.97	-43.25
29	32200.9	1576.7	1591	0.539	0.415	48.804	973.9	189.01	-39.77
30	32258.2	1714.1	1647	0.533	0.366	42.739	862.2	166.13	-37.7
31	32275.4	1472.7	1762	0.522	0.390	45.051	913.4	189.82	-40.58
32	32282.5	1467	1634	0.528	0.426	48.648	954.7	195.31	-38.88
33	32269.8	1762.8	1605	0.496	0.340	35.861	821	164.12	-35.54
34	32286.7	1609.8	1671	0.518	0.373	40.873	928.9	*177.82	*-36.02
35	32245.8	1635	1499	0.456	0.360	33.732	817	152.49	-28.57

NOTE: The asterisks (*) denote missing values for which the averages of existing values are indicated.

4.2 Analysis of the impact of various trailer undercarriage configurations on energy consumption

The tests required for analysis of the impact of the undercarriage on energy consumption were conducted on two roads in the Chénéville area. The total mass of the trucking rig was practically the same, with any slight variations due to the various components tested. Such variations were taken into account.

These tests were intended to

- study the laws of curvilinear movement combined with upgrades and downgrades, braking, acceleration and deceleration, i.e., to study the general laws of movement of an articulated multi-axle trucking rig;
- calculate, by means of a validated theory, the resistances to forward motion in curves and on grades with different trailer undercarriage configurations;
- establish, based on mean field measurements, the relationships between different trailer undercarriage configurations and energy consumption.

The general movement study was described in section 2. The analysis and the applications yielding practical results will be explained further on. We will deal here with the relationships between different trailer undercarriage configurations and energy consumption.

Let us start by examining the results of comparisons on the test rig comprising a trailer fitted with single tires and the same trailer fitted with double tires.

Constant-speed tests on a straight, level road yielded the following differences:
6.2% in favour of single tires on a tarred road;
1.4% in favour of single tires on a gravel road.

In Canada, vehicle loads and dimensions are legislated by provincial norms aimed at protecting bridges and road surfaces. For instance, the use of widely spaced rear axles on trailers allow for larger service loads but results in rig configurations that cause premature wear of the tires and wheel rims in curves. To offset this disadvantage, a drop axle was used as the first trailer axle. Since this is a sequential mechanism controlled from the cab, the drivers avoided problems by adjusting the drop course so that the axle barely touched the ground. This is the solution currently used on our roads.

This project therefore proposed to explore the difference between grouped and spread axles on the one hand and the combination of one or two self-steering front and rear axles on the other.

The differences in consumption for grouped and spread axles on the tarred road are 5.2, 4.3 and 3.9% at mean speeds of 30, 45 and 60 km/h, respectively. The difference on the gravel road is 2.6% smaller than on the tarred road at the same speed. Although appreciable, the differences in favour of grouped axles are not the only argument against the use of spread axles. Some equally important problems include fatigue and premature wear of the trailer chassis and undercarriage.

With a view to compliance with Canadian legislation on vehicle loads and dimensions, we investigated rigs comprising one self-steering axle mounted at the front and a second mounted at the rear. Analysis of the fuel consumption results shows differences of about 4% with one self-steering axle and 8% with two self-steering axles on the tarred road and differences of roughly 3.6% and 9.5%, respectively, on the gravel road. These results in favour of self-steering axles point up a considerable reduction in lateral resistances and thus in operating costs.

Since this study was not aimed at exploring the long-term operation, fatigue or durability of these axles, it would be good to contemplate studies along those lines. The introduction of self-steering axles under Canadian legislation on loads and dimensions would not only solve handling problems; it would also substantially reduce energy consumption on the road for equivalent total masses.

5. CALCULATIONS OF RESISTANCES TO FORWARD MOTION IN CURVES AND ON GRADES FOR DIFFERENT TRAILER UNDERCARRIAGE CONFIGURATIONS

The validated theoretical method of calculating the dynamic movement of an articulated multi-axle trucking rig, as well as data logged during testing at Chénéville on tarred and gravel roads, made it possible to simulate curves and grades and to establish the corresponding energy consumptions.

Figures 18 and 19 give, respectively, power and consumption for different vertical grades, as determined by road speed. For example, at a constant speed of 5 m/s on 3 and 12% grades, power amounts to 95 and 310 kW and consumption to 168 L/100 and 435 L/100 km, respectively. As was indicated earlier, the differences are tremendous, proving that those in charge of road maintenance and construction should examine the potential savings to be scored by reducing grades.

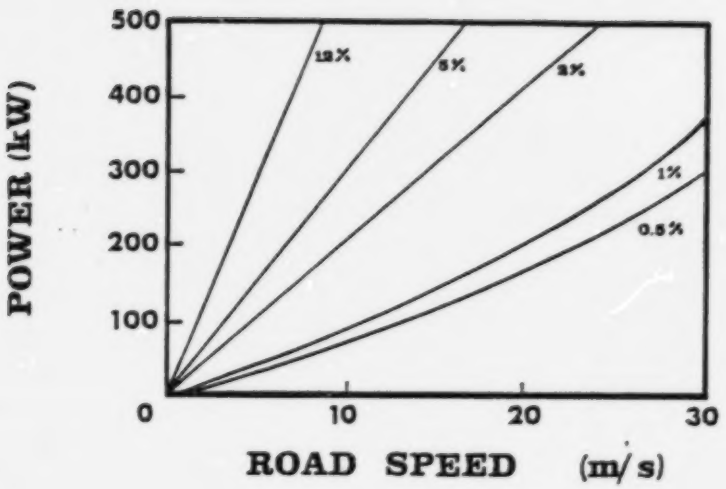

Figure 18 Power required to overcome grades and all resistance to forward motion on different grades as a function of road speed: 4 rigid axles, double tires, tire inflation pressure of 633 kPa

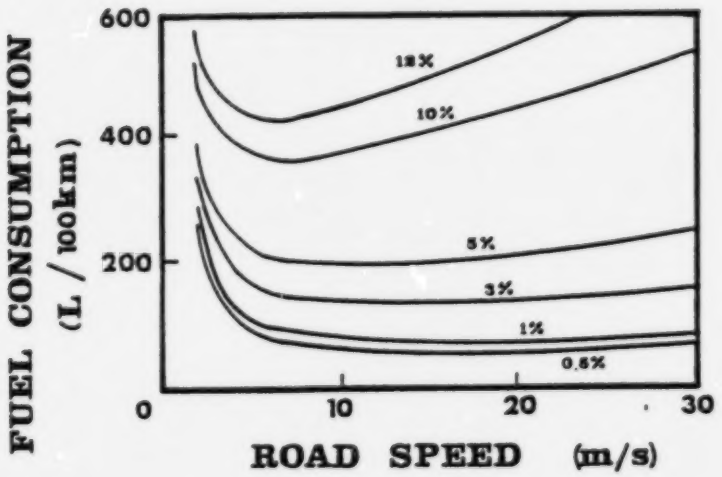

Figure 19 Total fuel consumption as a function of road speed for different grades: 4 rigid axles, double tires, tire inflation pressure of 633 kPa

Figures 20 and 21 give, respectively, power and consumption in L/100 km as determined by road speeds and curve radii. They show that the camber of a curve has considerable influence on fuel consumption. It is also obvious that these complementary resistances cause excessive fatigue and wear of all the undercarriage components of the rig.

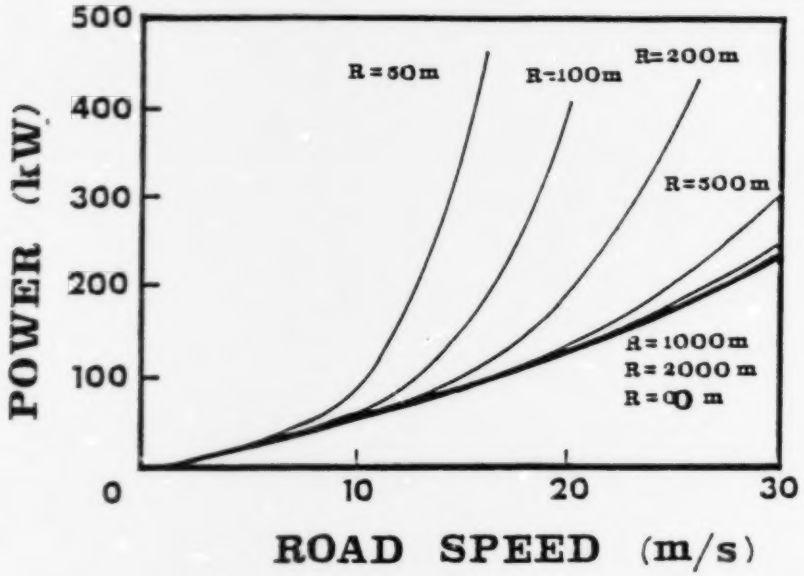

Figure 20 Power required to overcome resistance to forward motion for different curve radii as a function of road speed: grouped axles, double tires, tire inflation pressure of 633 kPa, tarred road

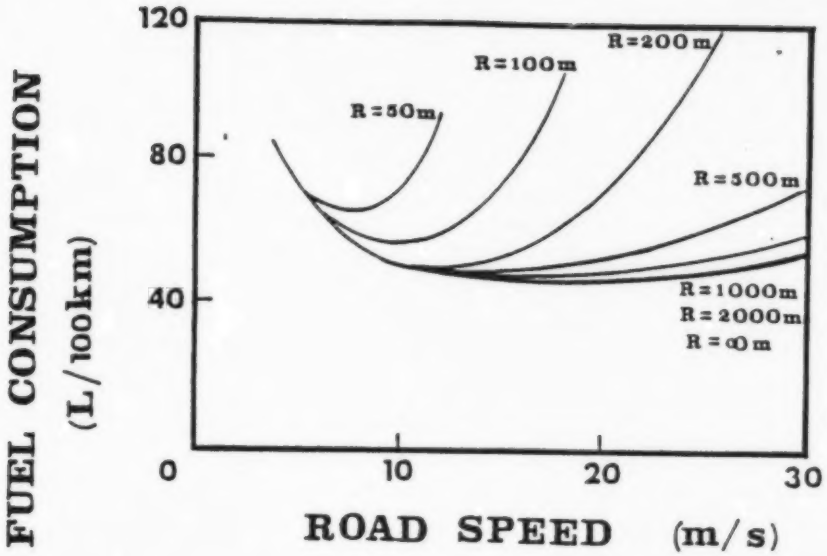

Figure 21 Fuel consumption for different curve radii as a function of road speed: grouped axles, double tires, tire inflation pressure of 633 kPa, tarred road

Table 3 presents selected figures calculated for two speed-radius combinations: 58 km/h and 200-m radius, and 80 km/h and 500-m radius. These combinations represent the average conditions encountered in the forest and on provincial roads. The table gives consumption data for various trailer undercarriage configurations. Numerous other configurations were calculated but are not given here. They are available on request.

Relative differences due to road surface are practically the same with four rigid spread axles or two rigid axles and two self-steering axles.

These tremendous differences point up the need for studies to determine the extent to which the surfaces and perhaps even the foundations of roads should be improved, without inordinate capital investment. This is particularly true for the forest industry.

Interestingly, if the forest industry opted to use two self-steering axles and single tires on gravel roads, consumption would be lower than with four rigid spread axles on tarred roads. The difference would be 2.6% less. In addition, the use of self-steering axles could simplify driving and reduce the excessive wear caused by widely spaced axles.

For general transport on tarred provincial roads, consumption with a trailer undercarriage equipped with two self-steering axles and double tires is 4.4% less than with four rigid grouped axles and double tires on the same roads. In other words, a trailer undercarriage with two self-steering axles is not only more fuel-efficient than one with four rigid grouped axles; it also prevents certain road-handling problems. On the other hand, enough space must be allowed for the operator to engage the self-steering axle locking system before moving in reverse. Under present conditions, Canadian truckers using rigid spread axles would incur fuel costs 6.1% higher when negotiating curves and equal, if not higher, costs owing to excessive wear of their rigs. The question is whether this is an economical means of complying with the legislation on loads and dimensions.

In the case of trailers with spread axles, it might therefore prove good to consider the use of self-steering axles.

In addition, the use of single wide-belted tires could spell fuel savings on the order of 15% (average for runs at load and no load)(1) and comparable savings in terms of vehicle wear.

To be certain that self-steering axles and wide-belted tires do indeed solve present road transport problems, one should first ascertain that the suspensions are compatible with this new technology. For example, independent pneumatic suspensions that allow for uniform load distribution over the axles and are controlled from the cab could possibly solve the problems posed by the use of self-steering axles and single tires. They could also allow for better control of the service load and improve no-load operation by deflating the suspension air chambers.

In the national interest, it is recommended here that studies aimed at the implementation of these new technologies be pursued and that the new legislation on vehicle loads and dimensions be amended accordingly.

(1) The difference in consumption for the no-load rig was estimated from previous tests.

Table 3 Test trailer undercarriage configurations and selected consumption data in L/100 km*

				(A)**	(B)***
4 Rigid axles (spread)	Tarred road	Single tires	490 kPa		
			633 kPa		
			773 kPa		
		Double tires	490 kPa		
			633 kPa	61.12	56.35
			773 kPa		
	Gravel road	Single tires	490 kPa		
			633 kPa	62.09	63.75
			773 kPa		
		Double tires	490 kPa		
			633 kPa	84.21	90.41
			773 kPa		
4 Rigid axles (grouped)	Tarred road	Single tires	490 kPa	49.16	47.17
			633 kPa	47.35	45.04
			773 kPa	44.50	42.08
		Double tires	490 kPa	57.49	55.68
			633 kPa	55.29	52.89
			773 kPa	51.02	48.32
	Gravel road	Single tires	490 kPa		
			633 kPa		
			773 kPa		
		Double tires	490 kPa	81.17	94.08
			633 kPa	75.47	84.99
			773 kPa	75.10	84.95
3 Rigid axles (spread) and 1 self-steering axle	Tarred road	Single tires	490 kPa		
			633 kPa		
			773 kPa		
		Double tires	490 kPa		
			633 kPa	56.30	53.49
			773 kPa		
	Gravel road	Single tires	490 kPa		
			633 kPa	58.62	61.65
			773 kPa		
		Double tires	490 kPa		
			633 kPa	78.98	85.92
			773 kPa		
2 Rigid axles (spread) and 2 self-steering axles	Tarred road	Single tires	490 kPa		
			633 kPa		
			773 kPa		
		Double tires	490 kPa		
			633 kPa	51.38	50.56
			773 kPa		
	Gravel road	Single tires	490 kPa		
			633 kPa	55.15	59.64
			773 kPa		
		Double tires	490 kPa		
			633 kPa	69.62	81.34
			773 kPa		

* Consumption derived by FERIC method, which uses coefficients calculated from field data as inputs for the conventional theory of road movement for an articulated multi-axle rig.

** (A) Road speed of 16 m/s (58 km/h) and curve radius of 200 m.

*** (B) Road speed of 22 m/s (80 km/h) and curve radius of 500 m.

6. SYNTHESIS OF SICMER CROSS-IMPACT ANALYSIS

The chief aim of this analysis was to devise a method of correlating driver's actions and machine performance on a given road, taking account of the operating environment as well.

Chapter 3 explains the relationships between the SICMER components, while chapter 4 analyses their cross-impacts based on data logged in the field. Knowledge of the laws governing these correlations is necessary to establish a SICMER study method.

A detailed analysis of these cross-impacts is presented in the report. The SICMER study method is illustrated on a nomogram of all the correlations derived from logged or calculated data (Fig. 10).

This nomogram can be used to follow the different paths originating at point A, at which the road speed and the dynamic demand are known.

Positioning point A in diagram D.8 makes it possible to deduce the possible transmission ratio(s), as well as the corresponding percentage(s) of pedal depression. Knowing the transmission ratio, D.10 can be used to find the engine rotating velocity. Using the pedal depression associated with the selected ratio, D.1, D.4 and D.5 can be used to establish engine torque, power and specific consumption, respectively. These last two items can be used in conjunction with D.11 to find the hourly rate of consumption simply by choosing the curve corresponding to the power read from D.4. The second path originating at point A determines the driver's pedal manoeuvres by using D.10 to establish the engine rotating velocity, then moving through D.12, D.13, D.7 and D.6 in that order.

A third path is used to establish hourly consumption from the characteristic engine diagram (D.3). D.10 is again used to determine the engine rotating velocity, and the path then continues through D.12, D.3 and D.14 in that order. This path has the advantage of disregarding the percentage of pedal depression, which was required with D.1, D.4 and D.5.

Interestingly, given adequate input data, any path on the nomogram can be covered in both directions. For instance, if the driver's choice of pedal depression, transmission ratio and engine rotating velocity are known, the hourly consumption can be established by starting from D.6 and continuing through D.7, D.13, D.3 and D.14. The road speed and specific dynamic demand of the road which that choice satisfies (or can satisfy) can be determined from D.6, D.7, D.13, D.12 and D.8.

Once laws have been obtained for a given articulated rig, certain elements can be fixed and the others varied in order to explore the impact of such variations on fuel consumption, power use and productivity.

The method of calculation developed was used to compare the impact of the rear axle ratios of the test truck on fuel consumption under identical driving conditions on the test road, i.e., same speeds, accelerations, braking, etc. Table 4 gives the comparative consumptions in terms of the rear axle ratio. It shows differences in consumption as high as 2.5% for the same driving technique, road, rig, speed and weight. This clearly indicates that transport companies can indeed improve fuel consumption through preliminary study of the choice of rear axle ratio as a function of operating conditions, particularly since this choice does not affect the cost of the truck.

Table 4 Fuel consumptions as determined by rear axle ratios on the FERIC test truck on the road between Chénéville and Montpellier (6.6 km)

Rear axle ratio	Section				Difference %
	1-900 m	900-1500 m	1500-5000 m	1-6600 m	
	mL	mL	mL	mL	
5.29	545	648	3033	5432.9	2.95
4.78	534.94	643	3003.13	5371.08	1.83
4.63	527.5	640	3011.23	5360	1.63
4.33	515	636	2960.44	5272.61	-
4.11	529.75	643.84	2952	5287.49	0.28
3.93	543	648	2978	5349	1.43
3.73	544	651	3009	5402	2.39

Given the large choice of transmissions on each truck model, they are very often ill-suited to the operating conditions, especially in terms of road profiles.

The method developed for the SICMER analysis made it possible to study the impact of three transmissions on fuel consumption. It is assumed here that all the machine and road parameters, with the exception of the transmission, are invariable. It is further assumed that the same road speed is maintained for all the test runs, meaning that time, distance and speed are the same for the different transmissions. In addition, the same power must be maintained to obtain the same speed at the same point on the road.

Table 5 gives the consumption obtained for the following transmissions:
- 8 ratio transmission (reference)
- 15 ratio transmission (used for previous tests)
- 7 ratio transmission.

Table 5 Cumulative fuel comsumptions over a distance of 6.6 km for selected transmissions

Type of transmission	Section				Difference %
	1-900 m	900-1500 m	1500-5000m	1-6600 m	
	mL	mL	mL	mL	
7 ratios	534	644	3032	5403	+0.6
15 ratios	514	635	2949	5258	-2.1
8 ratios	535	643	3003	5371	-
Infinitely variable transmission	495	635	2875	5113	-4.8

The 8-ratio transmission was used on the FERIC test truck and served as the basis for comparison. It yields fuel consumption 2.1% lower than with the 15-ratio transmission and 0.6% higher than with 7-ratio transmission.

Figure 22 gives the ratios used on a road section between 4500 and 6000 m. It shows that gears are shifted more frequently and are generally lower with the 15-ratio transmission. In this way, an attempt was made to offset the high rear axle ratio and approach 4.33 of this ratio, the value at which the best consumption was logged.

Lastly, we studied the hypothetical case of an infinitely variable transmission, which would provide for engine speeds corresponding to the minimum specific consumption for each pedal depression.

It was assumed in this scenario that only the transmission differed, the rig, road, speed and acceleration remaining the same.

Table 5 also gives consumptions for an infinitely variable transmission. It shows a 4.86% difference compared to consumption with the 8-ratio transmission. This is a considerable difference given that road performances are the same. Furthermore, automatic transmission and choice of engine rotating velocities will yield optimum engine use, regardless of the pedal position.

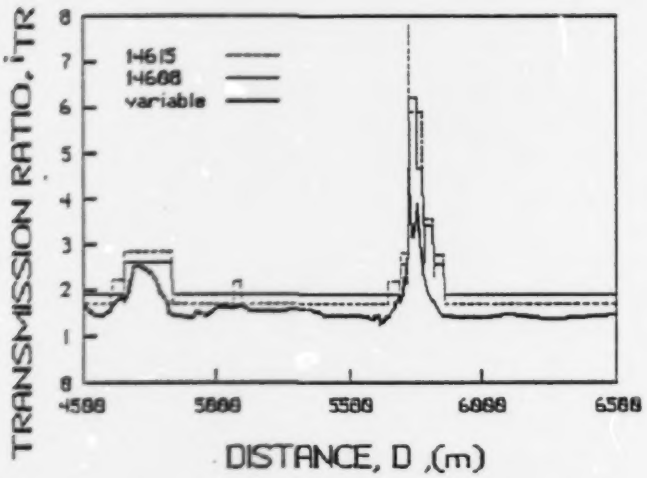

Figure 22 Ratios used for the 8- and 15-gear transmissions and the variable transmission (section between 4500 and 6500 m)

It is recommended that this highly promising solution be explored.

These few examples illustrate the possible applications of the method developed from the SICMER cross-impact analysis. Although these examples are confined to replacment of the "basic" truck elements (rear axle and transmission ratios), the method allows for analysis of the variation of more complex elements. It suffices in each case to determine the values of the parameters which are varied and to use the corresponding part of the calculation routine. This method will be used in future to establish a transport cycle simulation or optimization model comprising other necessary elements.

7. CONCLUSION

The results of this study bear out the accuracy of the system which we established. They incorporate the theories and methods developed in recent years and the results of tests undertaken during that time as part of FERIC's transportation research.

The original results exceed the set purpose and expectations. Not only do they open up new avenues in the complex area of cross-impacts between trucking rig components. They also point to possible improvements as regards energy conservation amd machine wear and performance.

Based on these findings, we recommend

- study of the use of self-steering axles on Canadian roads in terms of durability of the road surface, road safety, rig handling and durability of the components;
- study of the possibility of using self-steering axles while maintaining the same spacing and total weights and eliminating the lift of the first of the four trailer axles;
- continued studies aimed at the design of a universal model for transport cycle simulation and optimization;
- continued development of aids to economical driving;
- consideration of the introduction of an infinitely variable transmission with automatic (electronic) choice of engine RPMs corresponding to minimum fuel consumption;
- establishment of a plan for distribution of the results written up in this report.

In future, the SICMER analysis could serve as the basis for the design of a universal model for simulation of a vehicle on the road. Combined with data obtained on an existing road with the FERIC test truck or from an existing database, the model will be capable of realistic, reliable simulation or optimization of the SICMER system.